US005586870A

United States Patent [19]

Kawaguchi et al.

[11] Patent Number: 5,586,870

[45] Date of Patent: Dec. 24, 1996

[54] **BEARING STRUCTURE USED IN A COMPRESSOR**

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[21] Appl. No.: **277,347**

[22] Filed: **Jul. 19, 1994**

[30] **Foreign Application Priority Data**

Jul. 20, 1993 [JP] Japan 5-179394

[51] **Int. Cl.**^{6} **F04B 1/29**

[52] **U.S. Cl.** **417/222.2**; 417/269; 417/270; 417/362; 91/499

[58] **Field of Search** 417/269, 222.1, 417/222.2, 362, 270; 384/547; 92/12.2; 91/499, 506

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[57] **ABSTRACT**

A compressor includes a rotary drive shaft having a first end and a second end, both the ends being rotatably supported by opposing walls of a crank chamber defined in a casing and a swash plate tiltably mounted on the drive shaft and operably coupled with a piston in a cylinder bore. A rotation of the drive shaft results in a reciprocating movement of the piston for compressing refrigerant gas in a cylinder bore. A rotary member is secured to the first end which extends outside the casing. the rotary member supplies a power to the drive shaft from a power supply. A support member, disposed between the rotary member and the first end, protrudes outward from the casing. A first bearing, disposed between the support member and the rotary member, receives a thrust load and a radial load acting on the rotary member when the rotary member rotates. A second bearing, disposed between the casing and the second end, receives the radial load acting on the rotary member when the rotary member rotates.

17 Claims, 7 Drawing Sheets

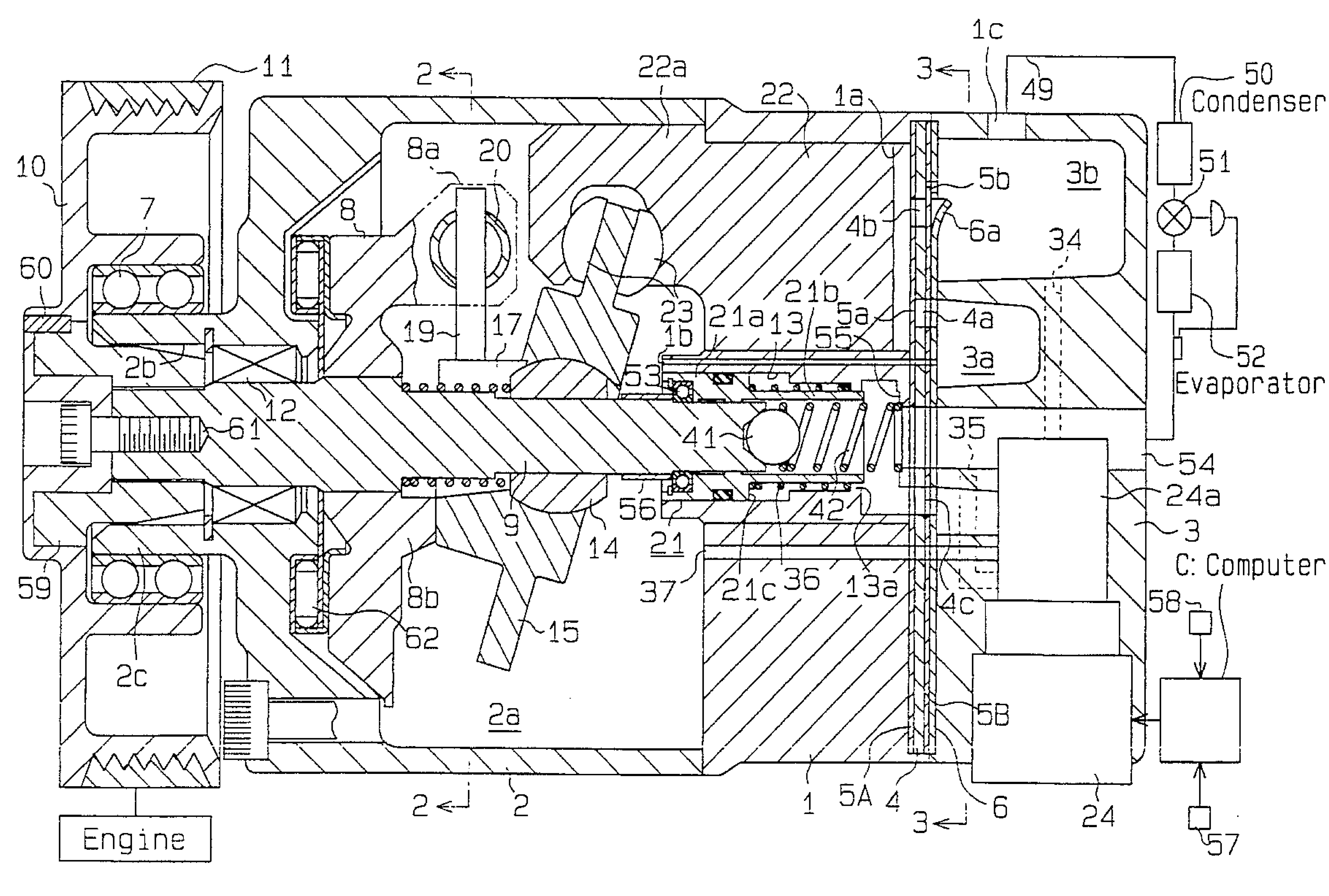


Fig. 1

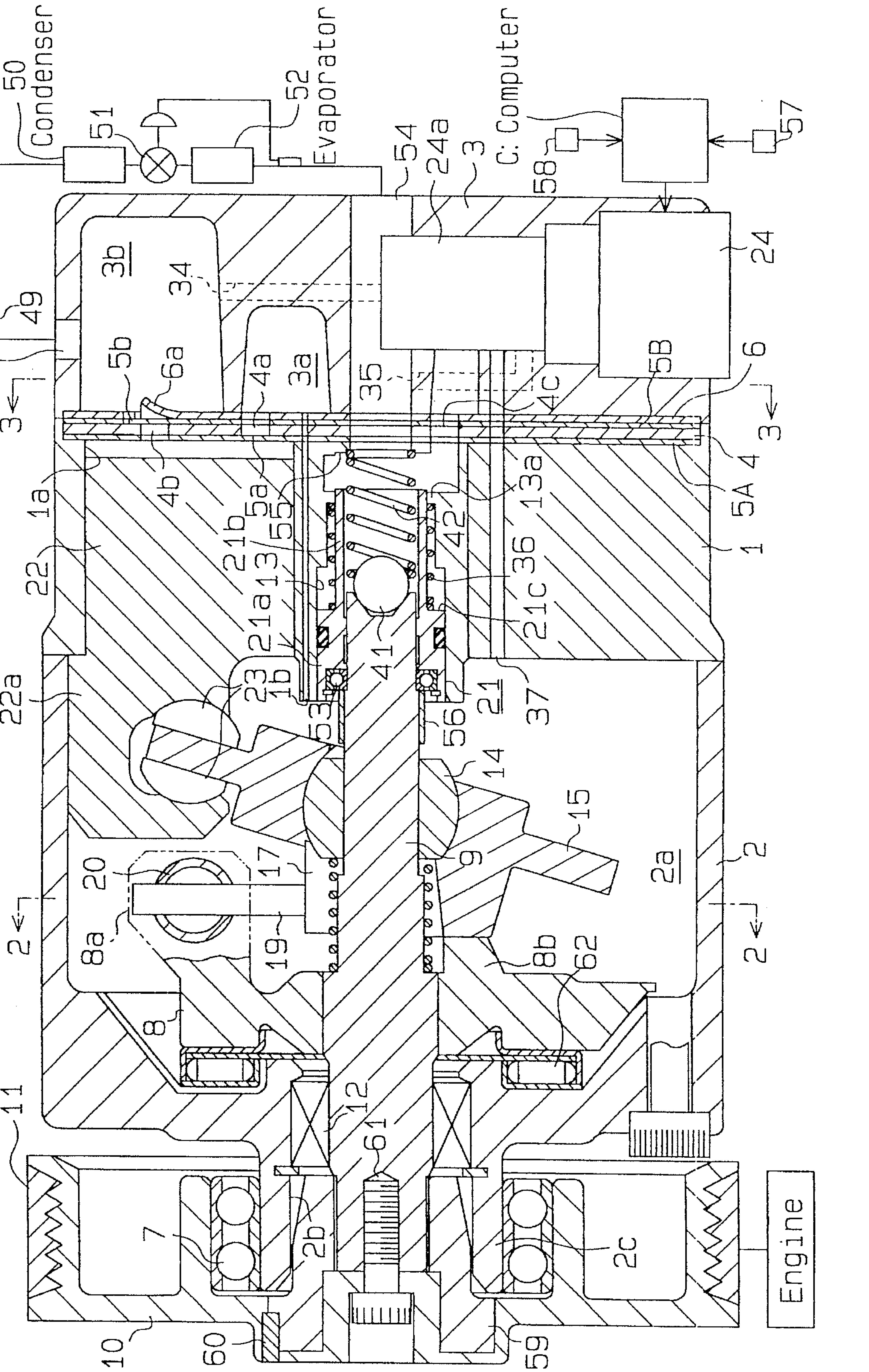


Fig. 2

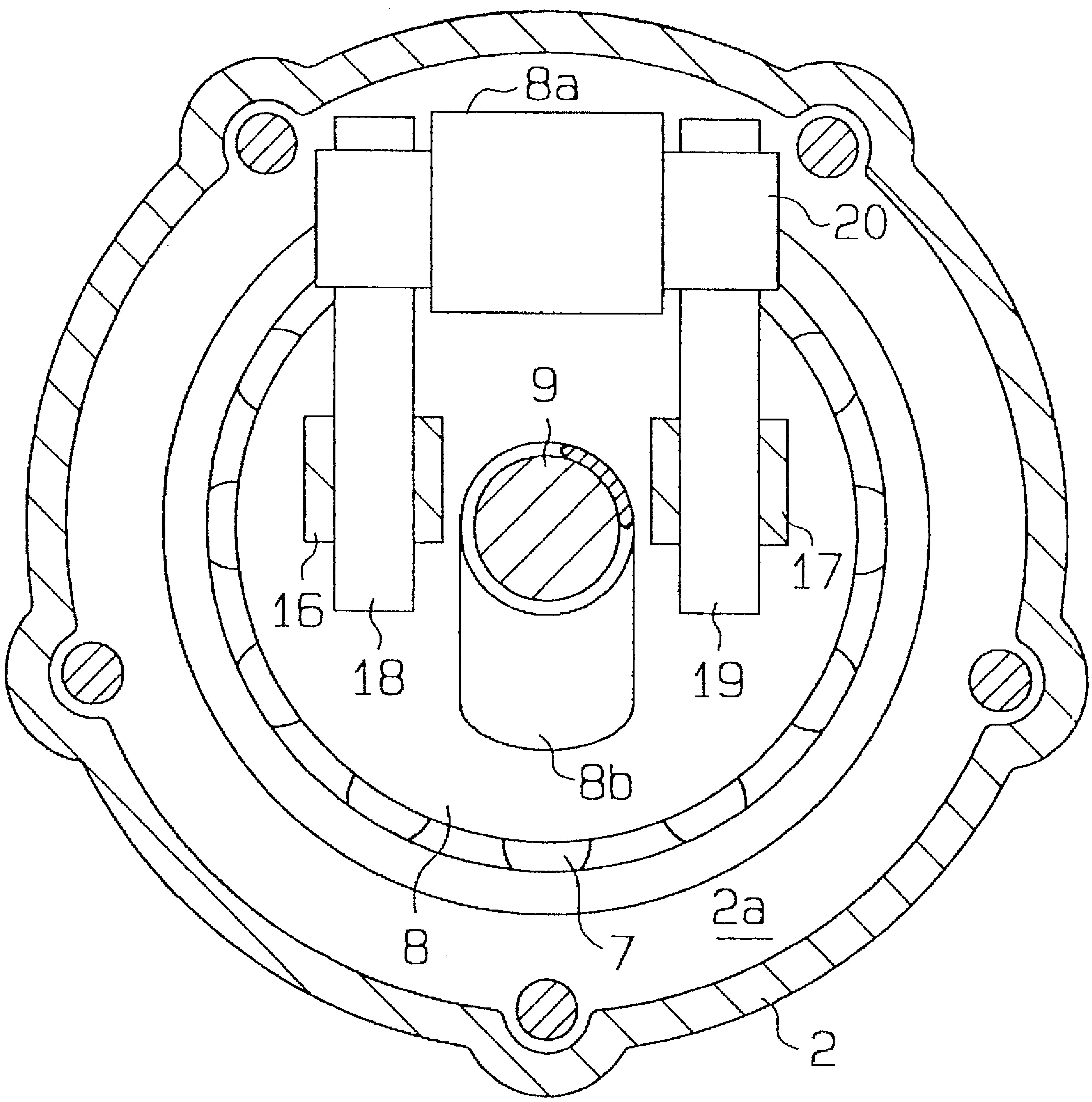
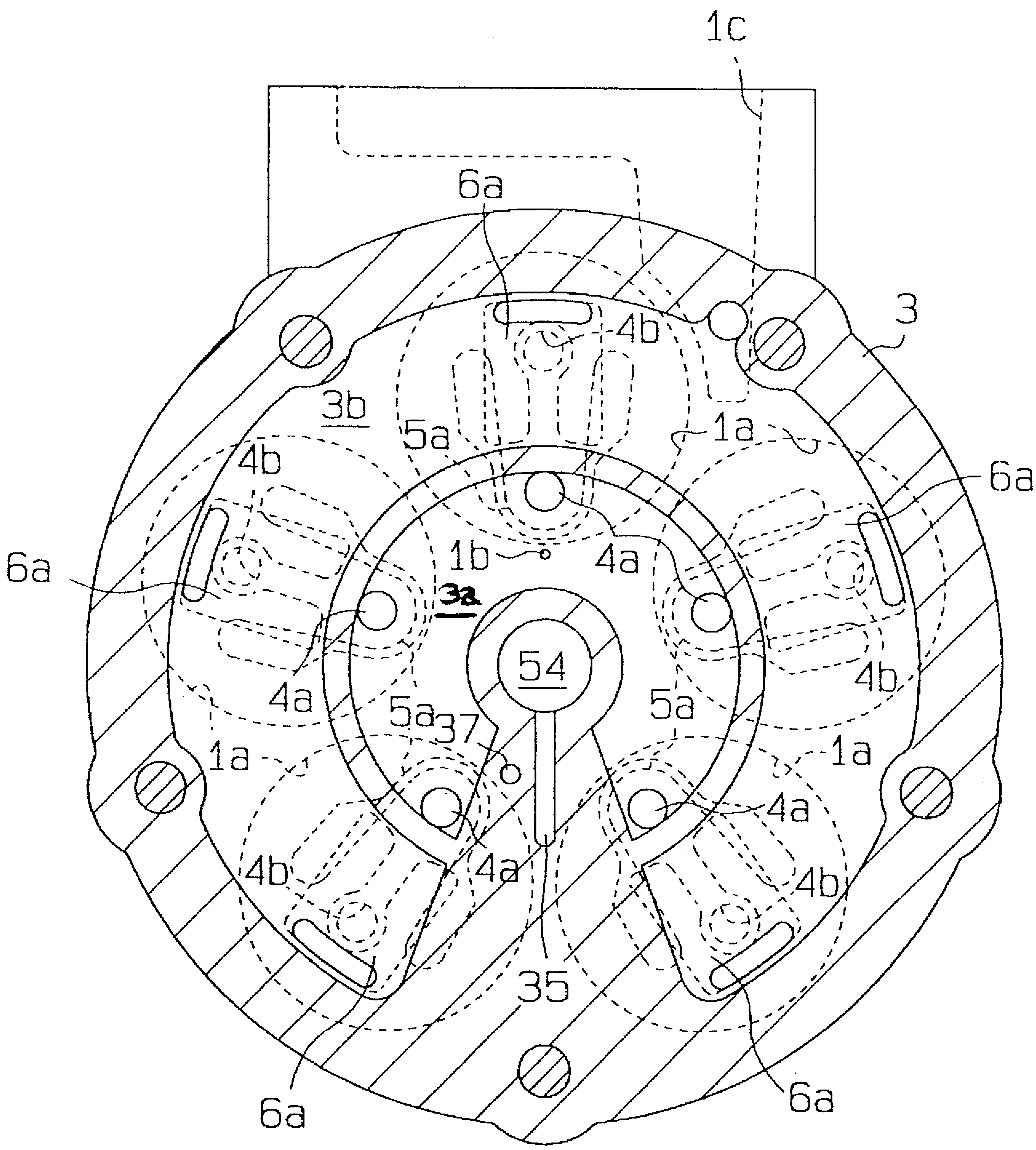


Fig. 3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4b
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4a
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4b
35
6a
6a

Fig. 4

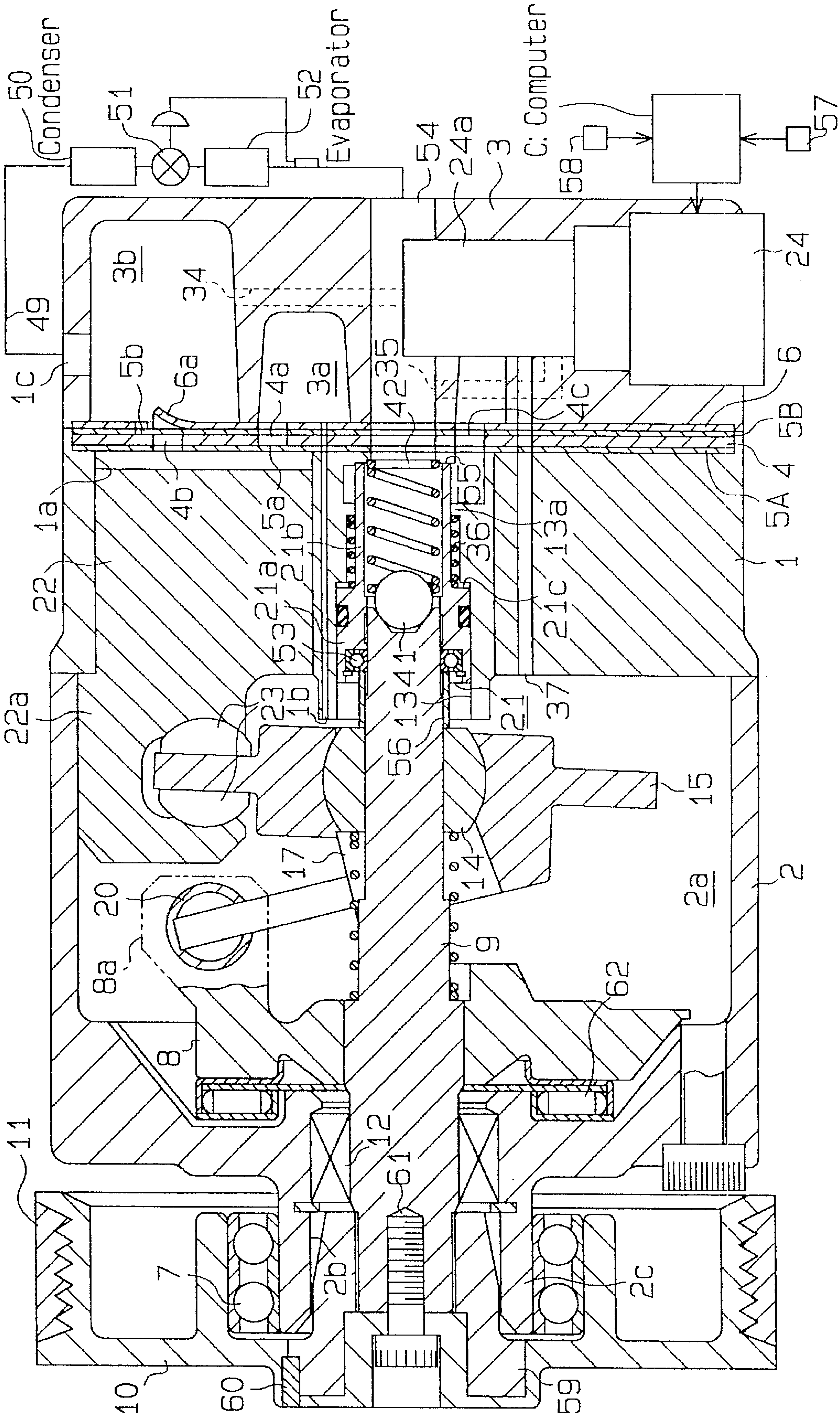
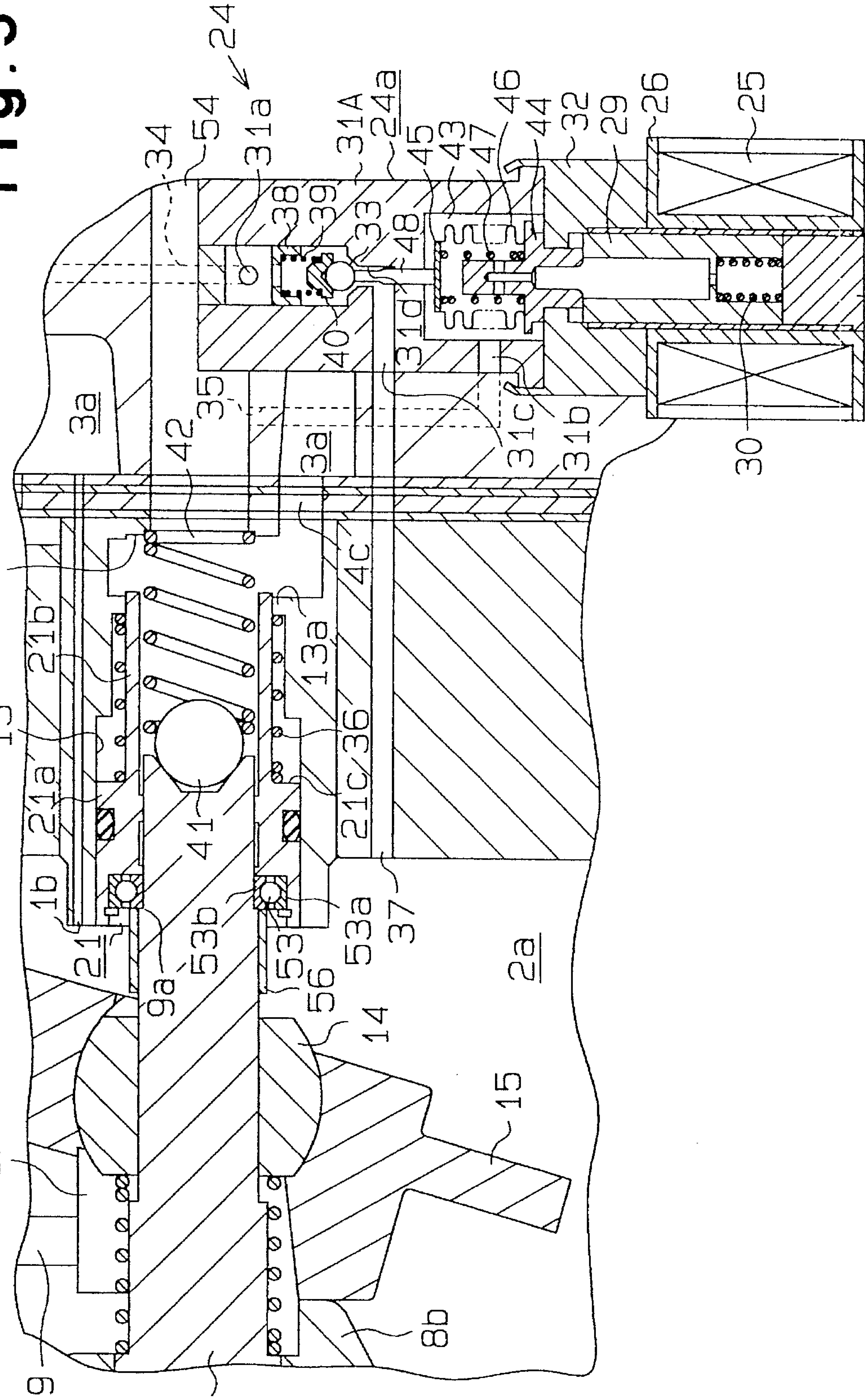


Fig. 5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40
31d
30
3a
35
42
3a
31c
31b
55
4c
21b
13a
13
36
21a
21c
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15
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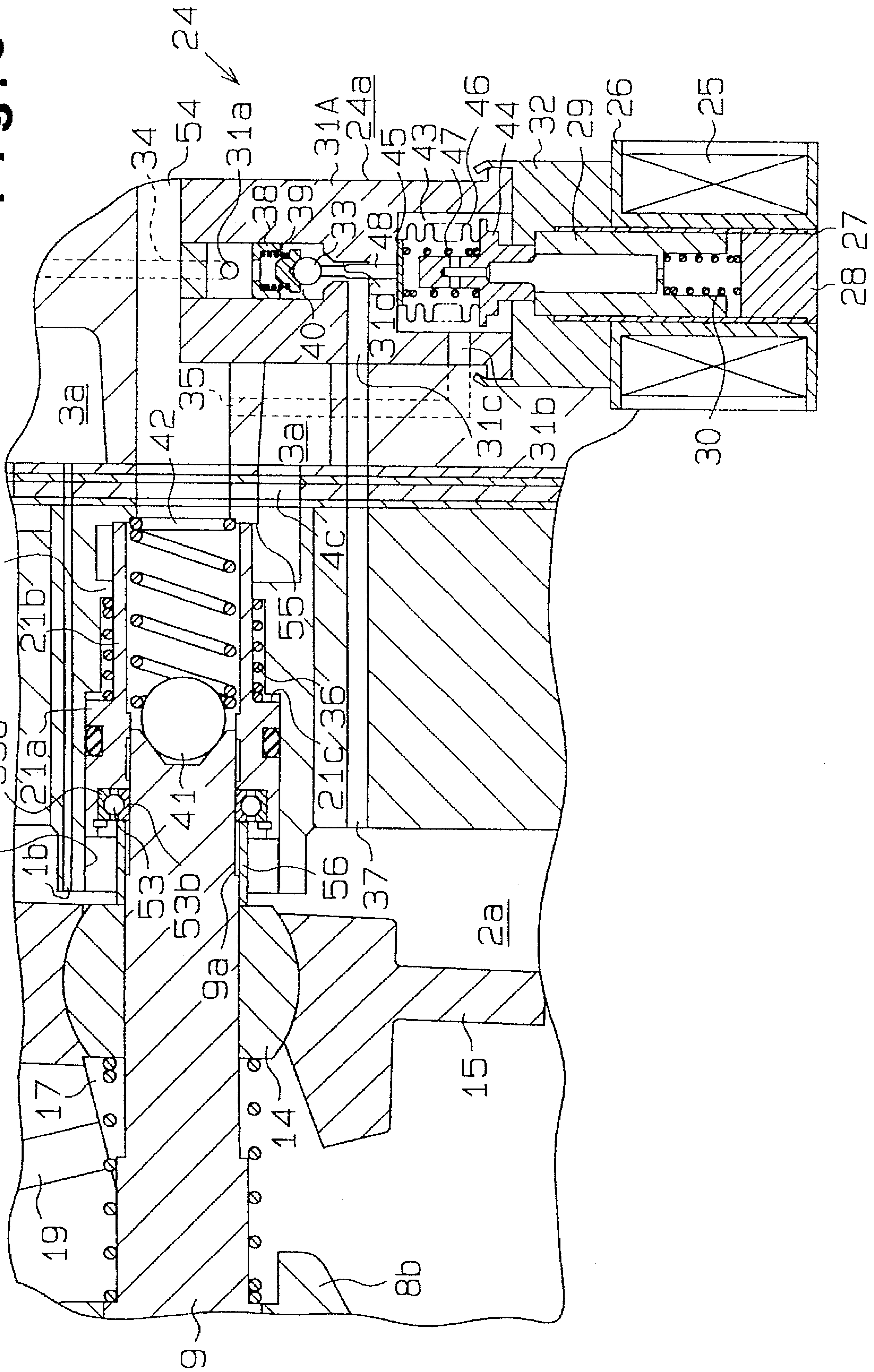
19
9
8b

Fig. 6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31d
40
27
28
30
31c
31b
35
3a
3a
42
4c
55
36
21c
13a
21b
21a
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53b
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56
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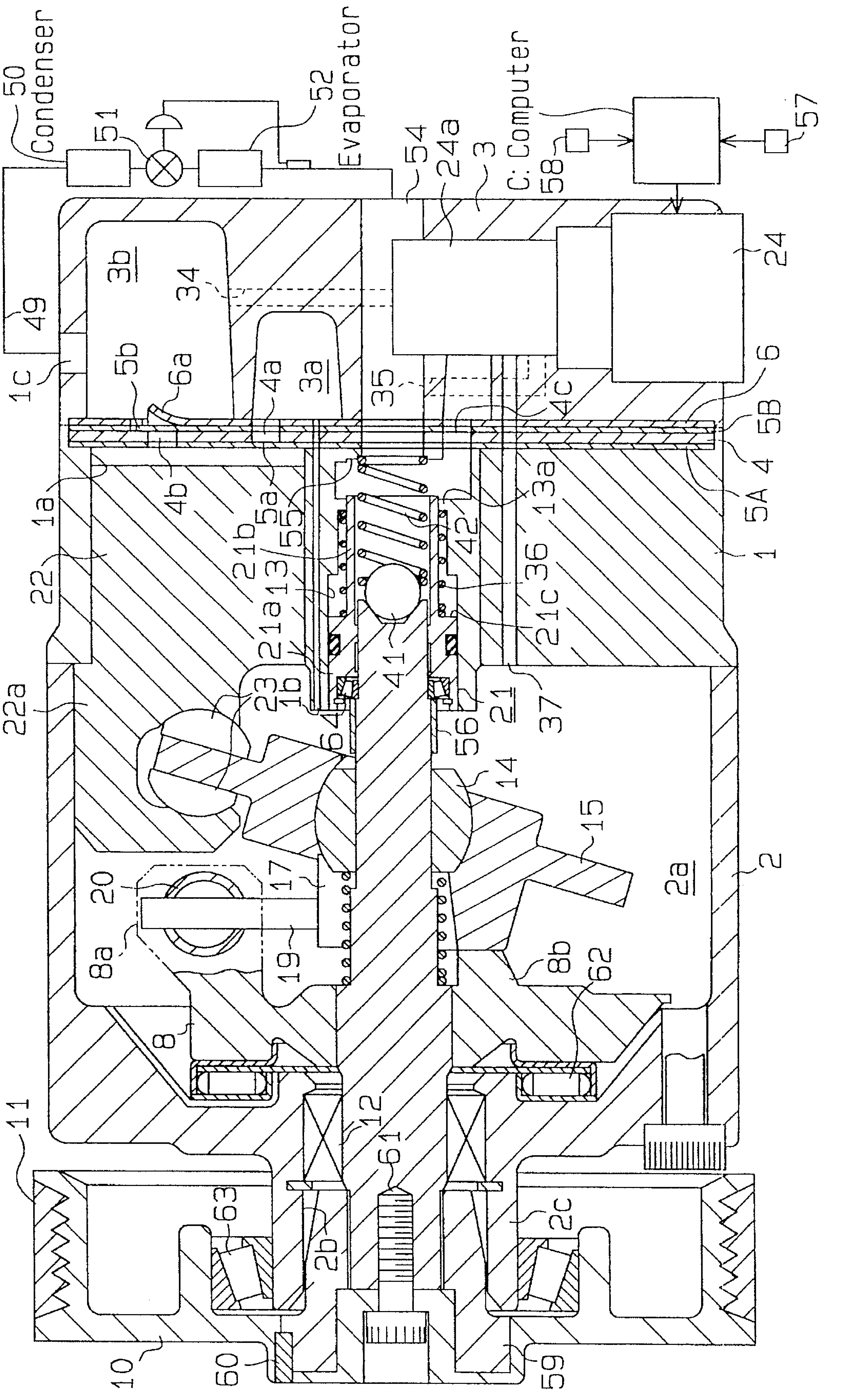
2a
9a
9
15
14
17
19
8b

Fig.7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52 Evaporator
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3
C: Computer
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57
24
3b
34
49
1c
5b
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3a
35
4c
6
5B
4
5A
1
1a
4b
5a
55
21b
13
21a
13a
42
36
21c
41
22
23
1b
64
21
56
37
22a
14
15
2a
2
20
17
8a
19
8b
62
8
12
11
61
63
2c
2b
10
60
59

BEARING STRUCTURE USED IN A COMPRESSOR

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to a shaft supporting structure in a compressor, in which a single-head piston is reciprocatable in a cylinder bore and a swash plate is tiltabley supported on a rotary shaft in crank chamber, and which controls the inclined angle of the swash plate based on the difference between pressure in a crank chamber and suction pressure.

2. Description of the Related Art

In general, compressor units used in automobiles, trucks and the like are used to supply compressed refrigerant gas to the vehicle's air conditioning system. To maintain air temperature inside the vehicle at a level comfortable for the vehicle's passengers, it is important to utilize a compressor whose displacement amount of the refrigerant gas is controllable. One known compressor of this type controls the inclined angle of a swash plate, tiltably supported on a rotary shaft, based on the difference between the pressure in a crank chamber and the suction pressure, and converts the rotational motion of the swash plate to the reciprocal linear motion of each piston.

In the conventional compressor, an electromagnetic clutch is provided between an external driving source, such as the vehicle's engine, and the rotary shaft of the compressor. Power transmission from the driving source to the rotary shaft is controlled by the ON/OFF action of this clutch. At the time the electromagnetic clutch is activated or deactivated, the clutch's action generates a shock generally detrimental not only to the compressor but also the overall driving comfort experienced by the vehicle's passengers. In addition, the integration of the clutch with the compressor increases the overall weight of the compressor.

The solve the above shortcomings, Japanese Unexamined Patent Publication No. 63-183277 discloses a variable displacement swash plate type compressor which does not employ an electromagnetic clutch for transmitting power from the external driving source to the compressor.

In this compressor, a pair of bearing members support a rotary shaft having a pulley secured thereto for power transmission from an external driving source. One of the bearing members is provided in a cylinder block while the other bearing member is provided in the housing which defines a crank chamber. The stable support of the rotary shaft results in the reduction of the vibration and noise of the compressor. This is achieved by increasing the distance between the bearing members. According to the structure in which the rotary shaft is supported in the housing, however, the distance between the bearing members cannot be increased unless the entire length of the compressor is increased.

SUMMARY OF THE INVENTION

It is an object of the present invention to provide a shaft supporting structure for a compressor that enhance the stable support of the rotary shaft in order to suppress vibration and noise without having to enlarge the size of the compressor.

To achieve the foregoing objects, according to the invention, a compressor is provided wherein a rotary drive shaft having a first end and a second end, both the ends being rotatably supported by opposing walls of a crank chamber defined in a casing and a swash plate tiltably mounted on the drive shaft and operably coupled with a piston in a cylinder bore. A rotation of the drive shaft results in a reciprocating movement of the piston for compressing refrigerant gas in a cylinder bore. A rotary member is secured to the first end which extends outside the casing. The rotary member supplies a power to the drive shaft from a power supply. A support member, disposed between the rotary member and the first end, protrudes outward from the casing. A first bearing, disposed between the support member and the rotary member, receives a thrust load and a radial load acting on the rotary member when the rotary member rotates. A second bearing, disposed between the casing and the second end, receives the radial load acting on the rotary member when the rotary member rotates.

BRIEF DESCRIPTION OF THE DRAWINGS

The features of the present invention that are believed to be novel are set forth with particularity in the appended claims. The invention, together with objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

FIG. **1** is a cross-sectional side view of an entire compressor according to an embodiment of the present invention;

FIG. **2** is a cross-sectional view along the line **2—2** in FIG. **1**;

FIG. **3** is a cross-sectional view along the line **3—3** in FIG. **1**;

FIG. **4** is a cross-sectional side view showing a swash plate at the minimum inclined angle;

FIG. **5** is a partially enlarged cross-sectional view of the compressor, showing a blocking body at an opening position;

FIG. **6** is a partially enlarged cross-sectional view of the compressor, showing the blocking body at a closing position and a solenoid in a de-excited state; and

FIG. **7** is a cross-sectional side view of an entire compressor according to another embodiment.

DESCRIPTION OF THE PREFERRED EMBODIMENT

A compressor according to a first embodiment of the present invention for use in an air conditioning system will now be described referring to FIGS. **1** through **6**. The compressor as described operates without a clutch between the external driving source and the compressor.

As shown in FIG. **1**, a front housing **2** is connected to the front end of a cylinder block **1**. A rear housing **3** is securely connected to the rear end of the cylinder block **1** via a valve plate **4**, valve forming plates **5**A and **5**B and a retainer forming plate **6**. The front housing **2**, cylinder block **1** and rear housing **3** constitute a casing of the compressor. The front housing **2** and the cylinder block **1** define a crank chamber **2*a***.

A rotary shaft **9** is rotatably supported between the front housing **2** and the cylinder block **1**. The front end of the rotary shaft **9** protrudes outside from the crank chamber **2*a*** through an opening **2*b*** of the front housing **2**. A cylindrical holder **59** is securely fastened on this protruding end portion. A pulley **10** is fitted on the holder **59**, with a key **60** intervening between the pulley **10** and the holder **59**. The key **60** inhibits the relative rotation between the pulley **10** and the holder **59**.

The pulley **10**, attached to the rotary shaft **9** by a screw **61**, rotates when coupled to a vehicle's running engine via a belt **11**. The pulley **10** rotates in the direction in which the holder **59** is fastened to the rotary shaft **9** by the screw **61**. Therefore, the pulley **10** and rotary shaft **9** rotate together, in one direction.

A support cylinder **2c** protrudes from the front end of the front housing **2** and surrounds the front end portion of the rotary shaft **9**. The driven pulley **10** is rotatably supported by the support cylinder **2c** via an angular contact ball bearing **7** which serves as a first bearing member. The support cylinder **2c** receives both the thrust load acting on the pulley **10** and the radial load via the angular bearing **7**.

A lip seal **12**, positioned between the support cylinder **2c** and rotary shaft **9** proximate to the front opening **2b**, prevents gas from leaking from inside the crank chamber to the outside along the rotary shaft **9**.

A drive plate **8** is secured on the rotary shaft **9**, and a spherical slider **14** is slidably supported on the rotary shaft **9**. A swash plate **15** is supported on the slider **14** in such a way as to be slidable in the axial direction of the rotary shaft **9**. Link pieces **16** and **17** are secured to the swash plate **15**. A pair of guide pins **18** and **19** are secured to the link pieces **16** and **17** as shown in FIG. **2**. A support arm **8a** protrudes from the drive plate **8**.

A support pin **20** is rotatably supported in the support arm **8a**, and extends perpendicular to the rotary shaft **9**. The pair of guide pins **18** and **19** are slidably fitted in the respective end portions of the support pin **20**. The interlocked action of the support pin **20** and the guide pins **18** and **19** permits the swash plate **15** not only to rock forward and backward along the axial direction of the rotary shaft **9** around the slider **14**, but to rotate together with the rotary shaft **9**.

A cylindrical space **13** is defined in the center portion of the cylinder block **1** in the axial direction of the rotary shaft **9**. A movable cylinder **21** is slidably retained in the space **13**, as shown in FIGS. **1**, **4** and **5**. The movable cylinder **21** has a large-diameter portion **21a** and a small-diameter portion **21b**. A spring **36** is interposed between a step portion **21c** at the outer surface of the movable cylinder **21** and a flange portion **13a** on the inner wall of the space **13**. The spring **36** urges the movable cylinder **21** toward the slider **14**.

The rear end portion of the rotary shaft **9** is fitted in the large-diameter portion **21a** of the movable cylinder **21**. A ball **41** is pressed against the rear end surface of the rotary shaft **9** by a spring **42**. The spring **42** serves to suppress the displacement of the rotary shaft **9** in the thrust direction.

A ball bearing **53** intervenes between the rear end portion of the rotary shaft **9** and the large-diameter portion **21a** of the movable cylinder **21**, as shown in FIG. **5**. The rear end portion of the rotary shaft **9** is supported by the inner wall of the space **13** via the ball bearing **53** and the movable cylinder **21**. The ball bearing **53** has both an outer ring **53a** secured to the inner surface of the large-diameter portion **21a** and an inner ring **53b** which is slidable on the surface of the rotary shaft **9**.

As shown in FIG. **5**, a step portion **9a** is formed in the outer peripheral surface of the rear end portion of the rotary shaft **9**, so that the movement of the inner ring **53b** toward the slider **14** is restricted by the step portion **9a**. Accordingly, when the ball bearing **53** abuts on the step portion **9a**, the movement of the movable cylinder **21** toward the slider **14** is inhibited.

A suction passage **54** is formed in the center portion of the rear housing **3** as shown in FIGS. **1** and **5**. The suction passage **54** is connected to the space **13**. A positioning surface **55** is formed around the opening of the suction passage **54** on the side of the space **13**. When the rear end surface of the small-diameter portion **21b** of the movable cylinder **21** abuts against the positioning surface **55**, the backward movement of the movable cylinder **21** is restricted and the communication between the suction passage **54** and the space **13** is blocked.

A transmission cylinder **56** is slidably provided on the rotary shaft **9** between the slider **14** and the ball bearing **53**. The transmission cylinder **56** has both a front end which can abut against the rear end surface of the slider **14** and a rear end that abuts against the inner ring **53b** of the ball bearing **53**.

As the slider **14** moves in a backward direction toward the movable cylinder **21**, the movable cylinder **21** abuts the transmission cylinder **56**, pressing it against the inner ring **53b** of the ball bearing **53**. The ball bearing **53** receives the load in the thrust direction as well as the load in the radial direction. Accordingly, the movable cylinder **21** is urged toward the positioning surface **55** against the force of the spring **36**, so that the rear end surface of the small-diameter portion **21b** abuts on the positioning surface **55**. The swash plate's inclined angle can thus be changed to a minimum value, i.e. slightly larger than zero degrees, as shown in FIG. **4**.

This occurs when the movable cylinder **21** comes to a closing position to block the communication between the suction passage **54** and the space **13**. The movable cylinder **21** moves between the closing and opening positions, in response to the slider **14**. It should be noted that the maximum inclined angle of the swash plate **15** is restricted by the abutment of a projection **8b** of the drive plate **8** on the swash plate **15**.

Single-head pistons **22** are retained in cylinder bores **1a**, formed through the cylinder block **1** so as to connect to the crank chamber **2a**. A pair of shoes **23** are slidably fitted over a neck **22a** of the piston **22**. The peripheral portion of the swash plate **15** is inserted between a pair of shoes **23** so that the end faces of both shoes **23** are in contact with both surfaces of the swash plate **15**. Accordingly, the rotational motion of the swash plate **15** is converted to the reciprocal motion of the piston **22** via the shoes **23**, causing the piston **22** to reciprocate in the associated cylinder bore **1a**.

As shown in FIGS. **1** and **3**, a suction chamber **3a** and a discharge chamber **3b** are defined in the rear housing **3**. A suction port **4a** and a discharge port **4b** are formed on the valve plate **4**. Suction valves **5a** and **5b** are formed on the valve forming plates **5A** and **5B** respectively. The backward returning action of each piston **22** allows the refrigerant gas in the suction chamber **3a** to push the suction valve **5a** open, and to enter each cylinder bore **1a** via its corresponding suction port **4a**.

When each piston **22** moves forward, the refrigerant gas that has entered each cylinder bore **1a** pushes each discharge valve **5b** to open and is then discharged into the discharge chamber **3b** through each discharge port **4b**. Each discharge valve **5b** abuts against each retainer **6a** on the retainer forming plate **6** so that the opening of the valve **5b** is restricted.

A thrust bearing **62** intervenes between the drive plate **8** and the front inner wall of the front housing **2**. The thrust bearing **62** receives a reactive force due to the compressed gas which acts on the drive plate **8** via the pistons **22**, shoes **23**, swash plate **15**, link pieces **16** and **17**, guide pins **18** and **19** and support pin **20** in accordance with the reciprocation of the pistons **22**.

The stroke of each piston **22** changes in accordance with the difference between the pressure in the crank chamber **2**a and the suction pressure in the cylinder bore **1**a. Thus, the inclined angle of the swash plate **15**, which affects the compression displacement, varies. The pressure in the crank chamber **2**a is controlled by a displacement control valve **24** that is attached to the rear housing **3**. The crank chamber **2**a is connected to the suction chamber **3**a via a pressure release passage **1**b that serves as a restriction to the flow of gas from the suction chamber **3**a to the crank chamber **2**a.

The suction chamber **3**a communicates with the space **13** via a port **4**c that is formed through each plate **4**, **5** and **6**. When the movable cylinder **21** moves to the closing position, the port **4**c is shut off from the suction passage **54**. It should be noted that the suction passage **54** is an inlet port through which refrigerant gas is supplied into the compressor. The movable cylinder **21** may shut off the communication between the suction passage **54** and the suction chamber **3**a at the downstream portion of the suction passage **54**.

The internal structure of the displacement control valve **24** will be described with reference to FIGS. **5** and **6**.

A bobbin **26** of the valve **24** supports a solenoid **25** on the outer periphery thereof. A guide cylinder **27** is fixed to the hollow portion of a bobbin **26**. A fixed core **28** is securely retained in the guide cylinder **27**. A movable core **29** is retained in the guide cylinder **27** in such a way that it moves to and away from the fixed core **28**. A spring **30** is disposed between the fixed core **28** and the movable core **29**. The movable core **29** is urged away from the fixed core **28** by the spring **30**.

A valve housing **31** is securely coupled to the bobbin **26** via a connecter **32**. A spherical valve **33** is disposed in the valve housing **31**. A first inlet port **31**a for introducing the discharge pressure therein, a second inlet port **31**b for introducing the suction pressure therein and a control port **31**c are formed in the valve housing **31**. The first inlet port **31**a communicates with the discharge chamber **3**b via a first passage **34**. The second inlet port **31**b communicates with the suction passage **54** via a second passage **35**, and the control port **31**c communicates with the crank chamber **2**a via a control passage **37**.

A return spring **39** and a valve seat **40** are disposed between a spring seat **38** and the valve **33** in the valve housing **31**. The valve **33** receives the force of the return spring **39** acting in the direction to close a valve hole **31**d.

A metal fitting **44** secured to the movable core **29** is retained in a suction-pressure sensing chamber **43**, which communicates with the second inlet port **31**b. The metal fitting **44** is coupled to a spring seat **45** by a bellows **46**. A spring **47** intervenes between the metal fitting **44** and the spring seat **45**. A rod **48** is secured to the spring seat **45**, with its distal end abutting on the valve **33**. The valve **33** opens or closes the valve hole **31**d in accordance with a change in suction pressure in the chamber **43**. When the valve hole **31**d is closed, the communication between the first inlet port **31**a and the control port **31**c is blocked.

An outlet port **1**c discharges the refrigerant gas from the discharge chamber **3**b into an external refrigerant circuit **49**, as shown in FIG. **1**. The outlet port **1**c is connected to the suction passage **54** by the external refrigerant circuit **49**. Provided on the external refrigerant circuit **49** are a condenser **50**, an expansion valve **51** and an evaporator **52**. The expansion valve **51** controls the amount of the refrigerant gas allowed in the circuit **49** in accordance with a change in gas pressure on the outlet side of the evaporator **52**.

A computer C controls the solenoid **25**. The computer C energizes the solenoid **25** in accordance with the ON action of the vehicle's air-conditioner switch **57** or the OFF action of the vehicle's accelerator switch **58**. The computer C also de-energizes the solenoid **25** in accordance with the OFF action of the switch **57** or the ON action of the switch **58**. FIG. **5** shows a state where the solenoid **25** is excited. With the solenoid **25** excited, the movable core **29** is attracted to the fixed core **28** against the force of the spring **30**, as shown in FIG. **5**.

With the solenoid **25** excited, the bellows **46** is displaced by an amount corresponding to the variation in suction pressure in the suction passage **54**. This displacement is transmitted to the valve **33** via the rod **48**. When the suction pressure is high, i.e. when cooling load is large, the opening of the valve **33** is reduced. This causes the amount of the refrigerant gas, which flows into the crank chamber **2**a from the discharge chamber **3**b via the passage **34**, the first inlet port **31**a, the valve hole **31**d, the control port **31**c and the control passage **37**, to be reduced. The refrigerant gas in the crank chamber **2**a flows to the suction chamber **3**a via the pressure release passage **1**b. This consequently reduces the pressure in the crank chamber **2**a. On the other hand, when the suction pressure in the cylinder bore **1**a is high, the difference between the pressure in the crank chamber **2**a and the suction pressure in the cylinder bore **1**a decreases. This causes the inclined angle of the swash plate **15** to increase as shown in FIGS. **1** and **5**.

When the suction pressure is low, i.e. when cooling load is small, on the other hand, the opening of the valve **33** increases. This causes an increase in the amount of the refrigerant gas flowing into the crank chamber **2**a from the discharge chamber **3**b, which increases the pressure in the crank chamber **2**a. When the suction pressure in the cylinder bore **1**a is low, the difference between the pressure in the crank chamber **2**a and the suction pressure in the cylinder bore **1**a increases. This consequently reduces the inclined angle of the swash plate.

When the suction pressure is extremely low, i.e. when no cooling load is applied to the compressor, the opening of valve **33** approaches the maximum. With the solenoid **25** de-energized, in accordance with the OFF action of the switch **57** or the ON action of the accelerator switch **58**, the movable core **29** moves away from the fixed core **28** due to the force of the spring **30**. This changes the opening of the valve **33** to a maximum, as shown in FIG. **6**. Under these circumstances, the refrigerant gas in the discharge chamber **3**b undergoes a rapid flows into the crank chamber **2**a. The consequent increase in the crank chamber's pressure quickly rises to a maximum, thus changing the inclined angle of the swash plate **15** to a minimum.

With a reduced inclined angle of the swash plate **15**, the slider **14** shifts toward the movable cylinder **21** and abuts on the transfer cylinder **56**. The transfer cylinder **56** pushes the inner ring **53**b of the ball bearing **53** backward. Thus, the transfer cylinder **56** is held between the slider **14** and the inner ring **53**b. The transfer cylinder **56** therefore can rotate together with the rotary shaft **9**. Due to the fact the transfer cylinder **56** abuts only on the inner ring **53**b of the ball bearing **53**, the rotary shaft **9**, rotates together with the slider **14**, the transfer cylinder **56** and the inner ring **53**b. No sliding motion occurs between the slider **14**, the transfer cylinder **56** and the inner ring **53**b.

When the transfer cylinder **56**, pressed against the ball bearing **53**, further moves toward the movable cylinder **21**, the movable cylinder **21** is pushed toward the positioning surface **55** so that the distal end of the small-diameter portion **21**b of the movable cylinder **21** approaches the positioning surface **55**. This design effects a gradual decline in the area through which the refrigerant gas passes, i.e., from the suction passage **54** to the suction chamber **3***a*. Accordingly, the amount of the refrigerant gas supplied into the cylinder bore **1***a* from the suction chamber **3***a* also decreases gradually, slowly reducing the discharge displacement or discharge pressure. This prevents large changes in the compressor's torque over short periods of time.

When the movable cylinder **21** abuts on the positioning surface **55**, the flow of the refrigerant gas from the external refrigerant circuit **49** to the suction chamber **3***a* stops. When the minimum inclined angle of the swash plate is other than at zero degrees, the gas is still discharged into the discharge chamber **3***b* from the cylinder bore **1***a*, effected even with the minimum inclined angle of the swash plate. The gas discharged into the discharge chamber **3***b* flows into the crank chamber **2***a* via the first passage **34**, the control valve **24** and the control passage **37**. The refrigerant gas in the crank chamber **2***a* flows into the suction chamber **3***a* via the pressure release passage **1***b*, and the refrigerant gas in the suction chamber **3***a* is fed into the cylinder bore **1***a* to be discharged into the discharge chamber **3***b*. At this time, it is apparent that there are pressure differences among the pressures in the discharge chamber **3***b*, the crank chamber **2***a* and the suction chamber **3***a*. The refrigerant gas in the compressor avoids flowing to the external refrigerant circuit **49**, preventing frosting in the evaporator **52**.

When the switch **57** turns on or when the accelerator switch **58** turns off in the state shown in FIG. **6**, the solenoid **25** is energized. This causes the movable core **29** to be attracted to the fixed core **28**. Then, the bellows **46** contracts due to the suction pressure in the chamber **43**, and the valve **33** closes the valve hole **31***d*.

When there are pressure differences among the pressures in the discharge chamber **3***b*, the crank chamber **2***a* and the suction chamber **3***a*, and when the valve **33** closes the valve hole **31***d*, the pressure in the crank chamber **2***a* decreases. This causes an increase to the inclined angle of the swash plate. This increase in the inclined angle causes the slider **14** to move away from the movable cylinder **21**.

The movable cylinder **21** follows the movement of the slider **14** due to the force of the spring **36**, so that the distal end of the small-diameter portion **21***b* is separated from the positioning surface **55**. This gradually increases the area through which the refrigerant gas passes, i.e., from the suction passage **54** to the suction chamber **3***a*. As a result the amount of the refrigerant gas flowing into the suction chamber **3***a* from the suction passage **54** increases slowly.

Accordingly, the amount of the refrigerant gas flowing into the cylinder bore **1***a* from the suction chamber **3***a* gradually increases, thus resulting in a slow increase in discharge displacement. As a result, the discharge pressure rises slowly, so that the torque in the compressor does not change greatly in a short period of time.

The rotary shaft **9** keeps rotating unless the external driving source stops. Stable support of the rotary shaft **9** is important to suppress the vibration and noise of the compressor. To ensure the stable support of the rotary shaft **9**, first, it is essential to increase the distance between the support positions of the pair of bearing members. According to this embodiment, the angular bearing **7**, which serves the first bearing member, is arranged in front of the front housing **2**. The ball bearing **53**, which serves the second bearing member, is located in the space **13** in the cylinder block **1**. This arrangement sets the distance between both bearings **7**, **53** greater than the distance in the prior art without increasing the entire length of the compressor.

As previously described, according to the conventional structure, the first bearing member is located within the front housing **2**. The distance between the first bearing member and the second bearing member cannot be increased unless the length of the front housing **2** is increased.

With the structure having the angular bearing **7** on the support cylinder **2***c* according to this embodiment, however, the length of the support cylinder **2***c* is substantially the same as the length of a portion of the rotary shaft **9** which protrudes from the front housing. It should be noted that this length is necessary to connect the pulley **10** to the rotary shaft **9** like in the prior art. Therefore, the distance between the angular bearing **7** and the ball bearing **53** becomes greater than the conventional one without increasing the length of the front housing **2**. This ensures a more stable support of the rotary shaft **9** than that provided by the prior art.

The radial load from the external driving source acting onto the pulley **10** via the belt **11** is entirely received by the support cylinder **2***c*. The thrust load acting on the pulley **10** is also received by the support cylinder **2***c*. Consequently, the pulley **10** is supported with enhanced stability. The rotary shaft secured to the pulley **10** is also supported by the angular bearing **7** with increased stability.

The smoothness in the sliding motion of the movable cylinder **21** may inhibit a rapid torque change in the compressor. The smoothness necessitates the stable support of the rotary shaft **9**. In an unstable supporting state the rotary shaft **9** vibrates, and the sliding motion of the movable cylinder **21** lacks smoothness. This makes it difficult to gradually change the flow of the supplied refrigerant gas. According to this embodiment, however, the rotary shaft **9** has stable support allowing the movable cylinder **21** to slide with a smooth movement.

The entire radial load applied to the driven pulley **10** via the belt **11** from the external driving source is received by the support cylinder **2***c*. The radial load is not applied to the rotary shaft **9**. This produces the following advantages.

In the conventional clutchless compressor, the refrigerant gas normally distributes lubricant oil to the individual bearing members supporting the rotary shaft inside the compressor. The lip seal is usually located in front of the front bearing member in order to prevent gas in the crank chamber from leaking along the rotary shaft to the outside. Such a lip seal necessitates increasing the length of the rotary shaft protruding from the front housing. Consequently, when a load applied to the pulley, a bending moment occurs on the rotary shaft with the front bearing member serving as a fulcrum point.

Usually the rotary shaft in the conventional compressor should be strong enough to endure the moment at least from the position where the front bearing member is located to the position where the pulley is secured. This typically necessitates increasing the diameter of the rotary shaft. Unfortunately, increasing the rotor's diameter causes an increase in the slide contact area between the lip seal and the rotary shaft. Since the lip seal is usually pressed against the rotary shaft by the pressure in the crank chamber, the increase in the slide contact area increases the sliding resistance between the rotary shaft and the lip seal. This contributes to a power loss and quickens the deterioration of the lip seal. Moreover, the increased diameter of the rotary shaft increases the peripheral velocity of the rotary shaft, further hastening the deterioration of the lip seal.

In the present invention, however, the radial load, applied to the pulley **10** from the external driving source, is entirely received by the support cylinder **2**c. The diameter of the rotary shaft **9** can be made smaller as compared with the conventional compressor, thus reducing the slide contact area of the lip seal **12** and the rotary shaft **9**. The decreased sliding area reduces sliding resistance between the rotary shaft **9** and the lip seal **12**, which in turn reduces the compressor's power loss and improves the durability of the lip seal **12**. The reduced diameter of the rotary shaft decreases the peripheral velocity of the rotary shaft **9**, further improving the durability of the lip seal **12**.

The present invention is not of course limited to the above described embodiment, and conical roller bearings **63** and **64** may be used as the first and second bearings as shown, for example, in FIG. **7**. The bearing **63** receives the thrust load which acts on the pulley **10** in the direction toward the cylinder block **1** from the front housing **2**. The thrust load which acts on the pulley **10** in the direction toward the front housing **2** from the cylinder block **1** is received by a thrust bearing **62** in the front housing **2**. The conical roller bearing **64** exhibits the same function as the ball bearing **53** in the above described embodiment.

Although the above described embodiments are directed to a clutchless compressor, the present invention can be applied to a clutchless compressor having no mechanism to prevent refrigerant gas from entering into the compressor from the external refrigerant circuit. Japanese Unexamined Patent Publication No. 63-183277 discloses such a clutchless type compressor. The compressor according to the '277 publication discloses a radial bearing used for the second bearing member to support the rotary shaft in the compressor.

The present examples and embodiments are to be considered as illustrative and not restrictive and the invention is not to be limited to the details given herein, but may be modified within the scope of the appended claims.

What is claimed is:

1. A compressor comprising a rotary drive shaft having a first end and second end, said first and second ends being rotatably supported by opposing walls of a crank chamber defined in a casing, a disk tiltably mounted on said drive shaft and operably coupled to a plurality of pistons, such that rotation of said drive shaft results in reciprocating movement of said pistons for compressing refrigerant gas in a corresponding plurality of cylinder bores;

said first end of said drive shaft extending outside said casing;

a rotary member continuously coupled to said first end of said drive shaft for directly driving said drive shaft from a driving source;

a rigid support member protruding outwardly from said casing and disposed between said rotary member and said first end of said drive shaft;

a first bearing disposed between said rigid support member and said rotary member for receiving a thrust load and a radial load acting on said rotary member when said rotary member is rotatably driven by said driving source; and

a second bearing disposed between said second end of said drive shaft and the adjacent one of said walls of said crank chamber for receiving a radial load acting on said rotary member when said rotary member is rotated; and

said drive shaft being rotatably supported by only said first and second bearings.

2. A compressor according to claim **1**, wherein said first bearing includes an angular contact ball bearing.

3. A compressor according to claim **1**, wherein said second bearing comprises

a cylinder slidably mounted on said second end of said drive shaft and reciprocatable along said drive shaft within a predetermined range; and

said compressor further comprising urging means for urging said cylinder toward said disk to move said cylinder to an extent based on the inclined angle of said disk.

4. A compressor according to claim **3**, wherein said second bearing further comprises:

a ball bearing disposed between said cylinder and said second end of said drive shaft;

a ball engageable with said second end of said drive shaft; and

a spring disposed between said casing and said second end of said drive shaft for urging said ball in the direction counter to said thrust load.

5. A compressor according to claim **3**, wherein said casing includes:

a cylinder block having a front section and a rear section;

a front housing secured to said front section of said cylinder block;

a rear housing secured to said rear section of said cylinder block; and

wherein said crank chamber is defined by said front housing and said cylinder block.

6. A compressor according to claim **5**, wherein said support member is formed integrally with said front housing.

7. A compressor according to claim **6**, further comprising a center portion and a peripheral portion formed with said rear housing, said center portion having a suction chamber and a suction passage for sucking the refrigerant gas into said suction chamber; and said peripheral portion having a discharge chamber; wherein said cylinder block defines a space for communicating said suction chamber with said suction passage and for accommodating said cylinder, wherein said cylinder selectively opens and closes said suction passage in accordance with said reciprocal movement of said cylinder.

8. A compressor according to claim **7**, wherein said rear housing has a valve for controlling the pressure of the refrigerant gas in said crank chamber in response to the pressure of refrigerant gas in said suction passage.

9. A compressor comprising a rotary drive shaft having a first end and a second end, said first and second ends being rotatably supported by opposing walls of a crank chamber defined in a casing, a disk tiltably mounted on said drive shaft and operably coupled to a plurality of pistons, such that rotation of said drive shaft results in reciprocating movement of said pistons for compressing refrigerant gas in a corresponding plurality of cylinder bores;

said first end of said drive shaft extending outside said casing;

a pulley continuously coupled to said first end of said drive shaft for directly driving said drive shaft from a driving source;

a rigid support member protruding outwardly from said casing and disposed between said pulley and said first end of said drive shaft, said support member being formed integrally with said casing;

a first bearing disposed between said rigid support member and said pulley for receiving a thrust load and a radial load acting on said pulley when said pulley is rotatably driven by said driving source; and a second bearing disposed between said second end of said drive shaft and the adjacent one of said walls of said crank chamber for receiving a radial load acting on said pulley when said pulley is rotated; and

said drive shaft being rotatably supported by only said first and second bearings.

10. A compressor according to claim **9**, wherein said first bearing includes an angular contact ball bearing.

11. A compressor according to claim **9**, wherein said second bearing comprises

a cylinder slidably mounted on said second end of said drive shaft and reciprocatable along said drive shaft within a predetermined range; and

said compressor further comprising urging means for urging said cylinder toward said disk to move said cylinder and said ball bearing to an extent based on the inclined angle of said disk.

12. A compressor according to claim **11**, wherein said second bearing further comprises:

a ball bearing disposed between said cylinder and said second end of said drive shaft;

a ball engageable with said second end of said drive shaft; and

a spring disposed between said casing and said second end of said drive shaft for urging said ball in the direction counter to said thrust load.

13. A compressor according to claim **11**, wherein said casing includes:

a cylinder block having a front section and a rear section;

a front housing secured to said front section of said cylinder block;

a rear housing secured to said rear section of said cylinder block; and

wherein said crank chamber is defined by said front housing and said cylinder block.

14. A compressor according to claim **13**, further comprising a center portion and a peripheral portion formed with said rear housing, said center portion having a suction chamber and a suction passage for sucking the refrigerant gas into the suction chamber; and said peripheral portion having a discharge chamber; wherein said cylinder block defines a space for communicating said suction chamber with said suction passage and for accommodating said cylinder, wherein said cylinder selectively opens and closes said suction passage in accordance with said reciprocal movement of said cylinder.

15. A compressor according to claim **14**, wherein said rear housing has a valve for controlling the pressure of the refrigerant gas in said crank chamber in response to the pressure of refrigerant gas in said suction passage.

16. A compressor comprising a rotary drive shaft having a first end and second end, said first and second ends being rotatably supported by opposing walls of a crank chamber defined in a casing, a disk tiltably mounted on said drive shaft and operably coupled to a plurality of pistons, such that rotation of said drive shaft results in reciprocating movement of said pistons for compressing refrigerant gas in a corresponding plurality of cylinder bores;

said first end of said drive shaft extending outside said casing;

a rotary member coupled to said first end of said drive shaft for directly driving said drive shaft from a driving source;

a rigid support member protruding outwardly from said casing and disposed between said rotary member and said first end of said drive shaft;

a first bearing disposed between said rigid support member and said rotary member for receiving a thrust load and a radial load acting on said rotary member when said rotary member is rotatably driven by said driving source;

a second bearing disposed between said second end of said drive shaft and the adjacent one of said walls of said crank chamber for receiving a radial load acting on said rotary member when said rotary member is rotated,

said second bearing comprising a cylinder slidably mounted on said second end of said drive shaft and reciprocatable along said drive shaft within a predetermined range; and

urging means for urging said cylinder toward said disk to move said cylinder to an extent based on the inclined angle of said disk.

17. The compressor according to claim **16**, wherein said rotary member is a pulley.

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